

PATENT SPECIFICATION

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COMPLETE SPECIFICATION

DRAWINGS ATTACHED

Improvements in or relating to Impeller Blading for Centrifugal Compressors

I, RUDOLPH BIRMANN, a citizen of the United States of America, residing at Highland Farm, Newtown, Bucks County, Commonwealth of Pennsylvania, United States of America, do hereby declare the invention, for which I pray that a patent may be granted to me, and the method by which it is to be performed, to be particularly described in and by the following statement:—

This invention relates to the shape, spacing and general arrangement of rotating impeller blades applicable generally to various types of centrifugal compressors.

In application No. 31,058/61 (Serial No. 941342) there is described a novel configuration of centrifugal impeller blading which is much more tolerant to a wide range of variation of the inlet whirl than hitherto-known impeller blading. Such a variation of the inlet whirl can be the result of changing the angular position of flow directing guide vanes that may be provided in the inlet passage to the impeller, or without such guide vanes or for a fixed position thereof, it may be merely the result of the compressor being called upon to handle at constant RPM widely varying flow rates. In either case, or a combination of these cases, the angle of the flow at the inlet to the impeller relative thereto changes over a wide range to the extent that flow separation can occur, and the compressor efficiency becomes very low.

It is therefore one of the objectives of the present invention to improve the compressor efficiency under conditions of off design point operation by preventing the separation of the flow from the surfaces of the impeller blading in the vicinity of the inlet under conditions of operation whereby the blade angle at the inlet differs substantially from the angle of the

approaching flow.

Another objective is to improve the compressor efficiency by preventing excessive growth of the boundary layers between inlet and discharge, particularly on the trailing side of the blades.

Still another objective is to achieve a more uniform distribution of the relative velocity within the blade channels particularly in the vicinity of the discharge from the impeller and thereby to minimize or eliminate mixing or momentum transfer losses in the diffuser so as to improve the diffuser efficiency and attain higher overall efficiencies of compression.

While the range of flow rates that a centrifugal compressor operating at constant RPM can handle is substantially larger than that which can be handled by an axial flow compressor, it is nevertheless quite narrow even if the centrifugal compressor is equipped with a vaneless diffuser for the purpose of increasing this flow range to the maximum possible extent. In such a centrifugal compressor the upper limit of the flow that can be handled is reached when the relative angle of the flow entering the impeller is approximately 15° larger than the blade inlet angle. If this is the case, the flow separates from the leading surface of the impeller blades in the vicinity of their inlets, and a flow contraction occurs within the initial portions of the blade passages with sonic velocity prevailing at the point of greatest contraction, thus preventing any further increase of the flow rate. This phenomenon is known as choking.

At the lower limit of the flow rate the flow angle relative to the blade is approximately 15° smaller than the inlet angle of the blades, and separation occurs on the trailing side thereof. This separation is

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accompanied by a phenomenon of flow instability known as pulsation, which prevents safe operation of the compressor at this and any lower flow rate.

5 To extend the operation of the compressor into its normally unstable range, the air entering the impeller can be given a
 10 prewhirl in the direction of rotation by means of adjustable inlet guide vanes, so as to cause the flow angle to coincide with the inlet angles of the blades. Such a positive prewhirl not only reduces the flow rate that can be handled by the compressor without pulsation, but it also reduces the
 15 pressure ratio that the compressor can produce under conditions of constant RPM. If a large reduction of pressure ratio is desired, a high prewhirl, and consequently a large positive vortex strength at the inlet
 20 must be used, which requires setting the inlet guide vanes at very small angles; or in other words, directing the absolute velocity component of the flow at the inlet at a small angle to the direction of rotation. If
 25 this is done, the direction of the inlet velocity relative to the impeller can no longer coincide with the blade inlet angles to the end that flow separation occurs on the leading side of the blades, bringing about such
 30 high flow losses that the compressor efficiency becomes very poor.

In addition to the aforescribed losses at the impeller inlet which rapidly increase with increasing misalignment of flow angle
 35 and blade angle, there are other major losses in conventional centrifugal compressors which result in a serious limitation of the efficiency that can be achieved. The considerable length (in the direction of the
 40 flow) of the passages formed by adjacent blades gives rise to a large growth of the boundary layers, particularly on the blade trailing sides, where the boundary layers can grow to such a thickness at the outer-
 45 most end of the blades that in effect the passage can only be partially filled with through flow and the rest of the passage area is occupied by the boundary layer in which large energy losses occur. In the
 50 filled portion of the passage the velocity distribution is highly uneven—directly adjacent to the pressure side (leading side) of each blade, the thru flow velocity is low and is much higher near the trailing
 55 side of the adjacent blade, falling off, however to zero or even to a negative value within the thick boundary layer on the latter side. Such an uneven velocity pattern within the individual blade passages and
 60 the fact that these patterns are separated layers gives rise to mixing (momentum transfer) losses that occur in the initial portion of the diffuser which receives the flow after it is discharged by the impeller pass-
 65 ages.

According to the invention there is provided a compressor comprising an impeller and a housing for said impeller, said impeller comprising a hub and blades carried thereby, characterized by the fact that each
 70 of said blades comprises two spaced sections of which the trailing section has an inlet edge projecting beyond the inlet edge of the leading section in a direction opposite the direction of elastic fluid flow.
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According to another feature of the invention, the discharge edge of the leading section projects beyond the discharge edge of the trailing section in the direction of elastic fluid flow.
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Further objects and advantages of the invention will become apparent from a study of the following specification when taken in conjunction with the accompanying drawings, in which:
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Figure 1 is an axial sectional view of a small compressor equipped with adjustable inlet guide vanes for varying the vortex strength at the impeller inlet, for the purpose of controlling the pressure ratio and
 90 the flow rate;

Figure 2 is an elevation looking at the inlet side of the impeller shown in Figure 1;

Figure 3 is a fragmentary view showing a
 95 development of a projection of the impeller on a cylinder coaxial therewith;

Figures 4A, 4B, 4C and 4D show four different types of impellers to which the invention is applicable; and
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Figures 5A, 5B and 5C illustrate diagrammatically the blades that are designed in accordance with the invention and that can be incorporated in any one of the impellers of Figures 4A, 4B, 4C and 4D, inclusive.
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Referring to Figure 1, air, which may be at atmospheric or other pressure, enters thru the annular entrance 44, and flows radially inward thru passages 42 to be
 110 diverted axially for entrance into the impeller blading. Vanes 46 are located in the passage 42 and are angularly adjustable to impart whirl to the entering air. These vanes are carried by individual shafts 48 journaled in the casting 2 and provided
 115 at their outer ends with individual sprockets 50, over which there is trained a chain for their simultaneous rotary adjustment.

After leaving the impeller the air is received by the radially extending vaneless
 120 diffuser 38 from which the compressed air is discharged into volute 39, and finally thru a tangential outlet (not shown) to the compressor discharge system.

Reference may now be made to the improved impeller construction which is particularly illustrated in Figures 1, 2 and 3. As has been indicated, in this type of variable pressure ratio compressor the incoming air has a quite large range of whirl
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imparted thereto with the result that the deviations of the angle of flow relative to the impeller from the inlet angles of the impeller blades greatly exceed the variations permitted by the use of conventional blade construction. The range of inlet vortex strength at the inlet, i.e., the product of the peripheral component of the absolute velocity of the entering air at any radius by the peripheral velocity of the inlet edge of the impeller at the same radius may vary from a negative value to a positive value which, in some cases, may exceed the value of the vortex strength at the impeller outlet. Vortex flow is set up in the radially extending vaneless passage 42 by the guide vanes so that the vortex strength at any radius of the inlet edge of the impeller blading is constant.

In order to prevent the flow from separating from the blade surfaces and to provide for smooth flow despite the variations indicated above, recourse is had in accordance with the present invention to induce the turning of the flow from the direction of its relative inlet velocity to the direction of the blade channels by means of boundary layer energization so as to set up smooth flow between the impeller blades throughout the passage of air from the entrance to the discharge of the impeller. In order to effect boundary layer energization it is necessary to have available either a pressure source or sink to provide deflection of flow. In accordance with the present invention advantage is taken of the fact that a pressure sink is available in a centrifugal impeller at the location of the trailing side of each impeller blade not far from its discharge end. Design of the impeller blades proceeds as follows:—

The inlet flow established at the entrance to the impeller by virtue of the presence of spin-producing guide vanes 46 and the considerable radial extent of the passage 42 is free vortex flow, i.e., flow characterized by a constant value of the product of the peripheral component of the absolute velocity by the radius from the axis of rotation. At the same time, there is highly curved flow adjacent to the impeller inlet in the meridional direction involving a high meridional component of flow velocity at the outside of the inlet eye and a relatively low meridional component of flow velocity near the impeller hub.

Assuming a start of design involving thin impeller blades, and designing for some chosen operating condition at which optimum efficiency is desired, the first consideration for design of the thin impeller blades is that of distribution of the inlet angles along the inlet edges to take the foregoing into account, and, if the radii of curvature of the approach passage are small,

this requires that the blade inlet angle at the tip be larger than the blade inlet angle at the hub. (The opposite is true in conventional impellers having, generally, blade elements which are radial throughout giving rise to helicoidal blade surfaces, with the inlet edges lying in a radial plane). Using desirable helicoidal blades having all elements radial, the inlet angle distribution problem is solved by extending the inlet edge portions of the blades as shown at 67 so that the inlet edges lie on a cone about the axis of rotation rather than in a radial plane. Axial sections of the approach flow passage are curved so that the radii of curvature of the outer and inner boundary walls of such sections have their centers lying on such a cone. Then the blade inlet edges are orthogonal to the approach passage walls providing for high efficiency.

Summarizing the foregoing, there are thus provided for a chosen operating condition, thin helicoidal blades which due to the conical location of their inlet edges, have the proper distribution of inlet angles along these edges with boundaries of the approach flow passage to which these edges are orthogonal. Actual thin blades, however, do not tolerate the considerable variation of vortex strength herein contemplated, providing good efficiency only in a narrow range of vortex strength.

Assuming that a thin blade of this type is designed it may be considered the center surface of a thick blade built up by the addition of material on both sides of this surface, the thickness being built up so that the maximum thickness is desirably not less than 10% and not more than 35% of the blade pitch measured at the same distance from the impeller axis. The maximum thickness should be located slightly downstream of the midpoint of the blade.

Next it may be considered that the thick blade thus provided is rounded at both its inlet and outlet edges to provide smooth airflow fairing. The result then would be a solid blade which, referring to Figure 3, would have a rounded inlet end as indicated by the dotted line at 116 and a rounded exit edge as indicated by the dotted line at 115. Between these rounded edges the surface of this hypothetical blade would on its concave side be the concave side of blade portion 68 and on its convex side be the convex side of blade portion 66.

Next, consider that this thick blade is made hollow, by removal of central material to provide a shell-like blade which has wall thicknesses corresponding to 66 and 68 in Figure 3. The next step in the design is to provide the openings at 117 and 118 as shown in Figures 2 and 3. The opening 117 is so provided that the respective blade sections 66 and 68 of a single

vane or blade terminate at their inlet edges 67 and 69 in such fashion that these edges at any radius terminate at a surface which is approximately normal to the air flow passage at the inlet. The result, as will be evident, is that the inlet edge 69 of the leading blade portion 68 will lie axially beyond the inlet edge 67 of the blade section 66 in the direction of air flow. It should be noted that the inlet edge 67, previously discussed, lies in a cone about the axis of rotation and is orthogonal to the flow, while the inlet edge 69 of blade section 68 is radial (or more nearly radial), being also orthogonal to the flow reaching it. A generally similar situation is provided at the outlet opening 118, so that the outlet edge 66 of the blade portion 66 terminates short of the outlet edge 68 of the blade portion 68 in the direction of air flow, the termination of the former being, therefore, at a radius from the axis of rotation less than the radius of the latter. The result is that each impeller blade has a passage within its boundary sections 66 and 68 which has an entrance at 117 and an exit at 118. It may be here noted that the flow of air relative to the impeller between the blade sections 66 and 68 of a single blade has an S configuration, while that between adjacent blades has only a single curvature.

The result of this construction is that, due to centrifugal action, a pressure sink exists at 119 producing a forced flow thru the opening 117 and between the blade sections with discharge at 118. The result is a forced turning of the flow at 117 which enforces if required a curvature of flow at the entrances of the blade passages because the boundary layers along the initial portion of the concave side of the blade section 66 are energized by being sucked away. It is near the blade inlet edge on this side of the blade section that, during adverse flow angle conditions, there would be separation of flow from the blade. The most adverse condition which would exist is that in which the spin component of the absolute velocity of the entering air would so exceed the peripheral velocity of the impeller that the relative velocity of the flow with respect to the blades might be more or less at right angles to the blades so that something of the order of a right angle turn of the flow would be required to permit the flow to enter smoothly the impeller passage between the blades. This condition may exist during such operating conditions as involve imparting a high whirl component to the air in the direction of rotation to cause the impeller to act as a turbine contributing to the drive of the rotor shaft. By reason of the induced flow of a portion of the air thru the passage between the vane sections 66 and 68 of a single blade despite the

adverse angle conditions, the flow between the blades is caused to curve so as to flow smoothly between them. Of course, under normal conditions for which the blades were originally designed the inlet angle relationships are proper, and under such conditions, the passages within the individual blades between their sections have no detrimental effect on the flow picture at the impeller inlet. They do have, however, a highly beneficial effect on the flow pattern at the outside diameter of the impeller. At this diameter the boundary layers that form on the trailing side of conventional blades reach their maximum thickness, which is often so great that the passages formed by adjacent blades become partially blocked by these zones of stagnation and turbulence, and the distribution of the discharge velocity along the periphery of the impeller is highly irregular. These conditions are greatly improved and the serious flow losses resulting therefrom are eliminated by the blades in accordance with the invention. The flow which is induced between the vane sections is discharged just short of the impeller outside diameter in such a manner that the boundary layers that would normally form on the trailing side of each blade are energized and blown away and the distribution of the discharge velocity relative to the impeller in the direction of its periphery becomes quite uniform.

Under conditions of extreme pressure ratio control wherein the impeller is required to operate at times as an expander or a turbine rather than a compressor, some further considerations of design are involved. For a turbine to show good efficiency it must operate with a rather high value of the ratio of peripheral blade speed with respect to the meridional velocity. Since the turbine action of the impeller described is concentrated in the inlet of the blades, the peripheral speed at the inlet must therefore be high to achieve fair turbine performance. This is at variance with the customary design approach for conventional impellers which need not have any turbine action. For these, the peripheral speed at the inlet is usually made as low as possible by using the smallest possible diameter of the inlet annulus to minimize the entrance Mach number and maximize the centrifugal effect to which the air is subjected when passing through the impeller passages. In accordance with the present invention, it is advantageous to design the impeller utilizing an inlet annulus having a relatively large diameter so that the mean square value of the inner and outer inlet diameters becomes approximately 55 to 60% of the discharge diameter of the impeller. At the same time, it is desirable that the radial extent of the blade at the

inlet should be no more than 30% of the mean inlet diameter.

While the impeller blading constructed in accordance with the invention as described is particularly advantageous if used in conjunction with pressure ratio and flow rate control by means of adjustable inlet guide vanes, it can also be used with great advantages for other purposes; for example, to improve the efficiency and to increase the useful operating range of any type of centrifugal compressor. It should be noted that the procedure given above applies to the design of the blading for any one of the four types of impellers depicted by Figure 4A to 4D. These four types are distinguished from each other by the direction of the inlet to and the discharge from the impeller as viewed in a meridional view and are accordingly known, respectively, as the radial inlet-radial discharge, the axial inlet-radial discharge, the axial inlet-diagonal discharge, and the axial inlet-axial discharge types.

Another common way of classifying centrifugal impellers has to do with the direction of the flow discharging from the impeller (relative thereto). Impellers are made to have a backward, 90° or forward discharge. The previously described design procedure for the new type of blades applied to these three types of impellers yields the shapes and arrangements of the blades shown by Figures 5A, 5B and 5C. These figures show diagrammatically the intersection of the blades of any one of the impellers shown by Figures 4A to 4D with the mean surface of revolution between the boundaries of the flow passages extending from the impeller inlet to the discharge.

Despite the difference of the three blade configurations shown by Figures 5A to 5C, their consistency with the basic description presented in connection with Figures 1, 2 and 3 can easily be recognized. In every case, the two sections of each blade define a fairly thick airfoil, the mean line of which is curved from the required inlet angle to the required discharge angle of the centrifugal impeller. This airfoil is opened at the inlet (leading edge) end and the outlet (trailing end) for the passage of boundary layer energizing air in such a fashion that the trailing side blade section is singly curved and extends from the impeller inlet face to a distance short of the impeller discharge annulus and the leading side blade section which overlaps the other section over a considerable distance in the direction of the flow, is doubly curved and extends from a distance downstream of the inlet all the way to the impeller discharge annulus.

The functioning of all the various blade configurations embodying the invention is

also uniformly the same, as follows.

The boundary layer energizing air which is induced to flow between the blade sections sucks the boundary layer away from the exposed leading side of the trailing blade section, preventing flow separation at this point of the inlet under conditions of the operation wherein the angle of flow approaching the inlet is larger than the angle of the blade. Under all conditions of operation, this boundary layer energizing air blows away the thick boundary layer that would form otherwise on the blade trailing side in the vicinity of its discharge end, and at the same time this flow of air between the blade section causes the discharge velocity between the blades to be evened out and to become quite uniform. This uniformity results in the achievement of a high efficiency of the conversion into pressure (in the diffuser) of the kinetic energy imparted to the air by the impeller.

WHAT I CLAIM IS:—

1. A compressor comprising an impeller and a housing for said impeller, said impeller comprising a hub and blades carried thereby, characterized by the fact that each of said blades comprises two spaced sections of which the trailing section has an inlet edge projecting beyond the inlet edge of the leading section in a direction opposite the direction of elastic fluid flow.

2. A compressor according to claim 1 in which the discharge edge of the leading section projects beyond the discharge edge of the trailing section in the direction of elastic fluid flow.

3. A compressor according to either claim 1 or claim 2 in which said blades and hub provide passages between the blades which direct elastic fluid at the discharge ends of the passages to have a substantial radial outward component of flow.

4. A compressor according to any of the preceding claims in which the inlet edges of said trailing sections lie in a cone concentric with the axis of rotation and having its vertex pointing opposite to the direction of elastic fluid approach to the impeller.

5. A compressor according to any of the preceding claims in which said sections of each blade define an S-passage for flow of elastic fluid between the sections.

6. A compressor according to any of the preceding claims in which the inlet eye to the impeller has radial dimensions such that the mean square of the inlet and outlet diameters of the eye is approximately in the range of 55% to 60% of the discharge diameter of the impeller.

7. A compressor according to claim 6 in which the radial extent of the inlet eye is no more than 30% of the mean inlet diameter.

8. The combination of a compressor

according to any of the preceding claims with means for directing elastic fluid to said blades, the last mentioned means providing an annular flow passage and angularly adjustable vanes at the outer portion of said passage and substantially spaced from said impeller blades, thereby to provide inward vortex flow of the elastic fluid in its approach to the impeller.

9. A compressor substantially as here- 10
inbefore described with reference to the
accompanying drawings.

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#941343 COMPLETE SPECIFICATION
2 SHEETS

This drawing is a reproduction of
the Original on a reduced scale.

SHEETS 1 & 2

(BIRMANV) - Nov. 1963 - 11

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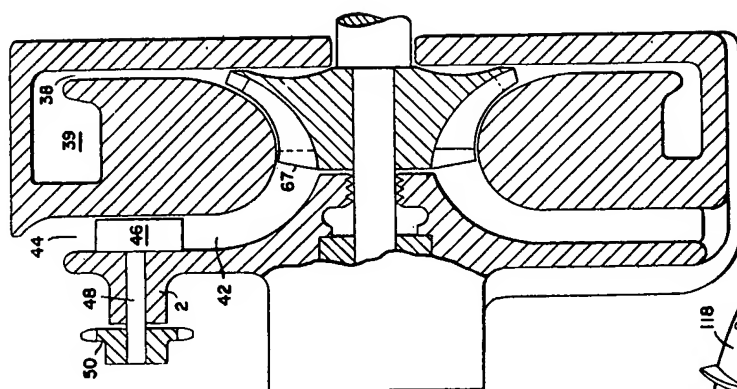


FIG. 1.

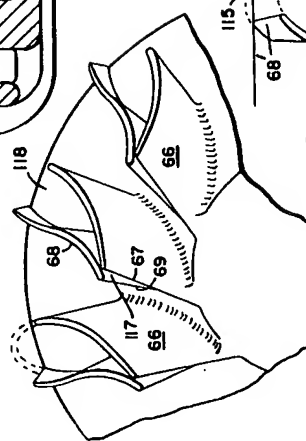


FIG. 2.

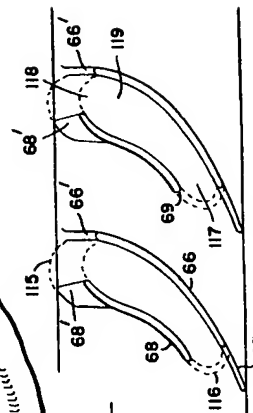


FIG. 3.

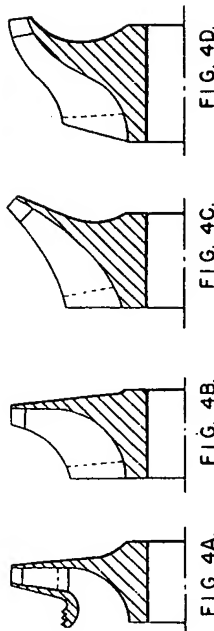


FIG. 4A.

FIG. 4B.

FIG. 4C.

FIG. 4D.

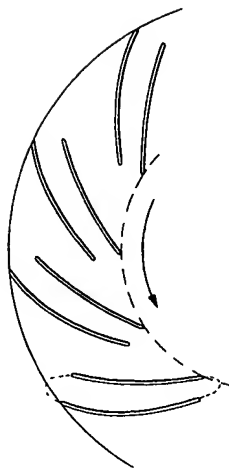


FIG. 5A.

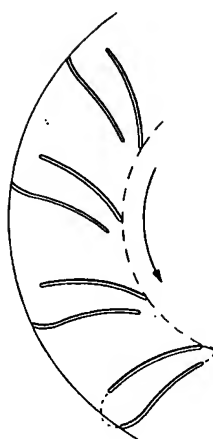


FIG. 5B.



FIG. 5C.